Thread control device for a textile machine, in particular for a shedding device

Technical Field

The invention relates to a thread control device for a textile machine, in particular for a shedding device, according to the preamble of claim 1.

Prior Art

Large numbers of thread control devices for textile machines are known. The nearest prior art according to WO 97/08373 discloses a thread control device which is designed with a drive and with a return device for a thread guide member. The thread guide member is in this case moveable in one direction of movement by means of the positively designed drive and in the opposite direction of movement by means of a nonpositive and pneumatically designed return device acting counter to the positive drive.
The pneumatic return device has a cylinder/piston assembly, the cylinder chamber of which is designed with an excess pressure valve and with a non-return valve which is connected to a compressed gas source. The gas pressure in the cylinder chamber is in this case set as a function of the operating state of the textile machine. For example, in a creep-speed phase, the gas pressure is kept lower than in a high-speed phase, so that the electric motor can furnish the necessary power for overcoming the load occurring as a result of the compression of the cylinder chamber. In a high-speed phase, the electric motor delivers sufficient power, so that the gas pressure can be increased further in order to prevent a roller on a cam disk of the positive drive from lifting off. Furthermore, the cylinder chamber may be designed with a manually actuable pressure relief valve, in order,

when the textile machine is being set up, to minimize the resistance occurring as a result of the compression in the cylinder chamber.

The above solution has the disadvantage that the gas pressure in the cylinder chamber has to be adapted to a respective operating state. This necessitates a complicated pressure control device for setting the gas pressure of the cylinder chamber, which requires pressure reducing valves and opening valves for activating each cylinder chamber. Moreover, a complicated electronic control of the valves is necessary in order to adapt the pressure in the cylinder chambers to a respective operating state.

To lubricate the cylinder/piston assembly, oil drops onto the piston, for example from above, and, due to hydrodynamic effects, enters the cylinder chamber despite a permanent excess pressure in the latter. The oil which has accumulated in the cylinder chamber may persistently disrupt the operation of the thread control device, since it reduces the air volume in the cylinder chamber to an indeterminate level, thus leading, during operation, to higher incalculable compression pressures in the chamber. In an extreme case where a large part of the cylinder chamber is filled with oil, it is no longer possible for the cylinder to move and further operation of the textile machine would lead to considerable damage.

In an improved embodiment of the pneumatic return device described in WO 97/08373, therefore, the valve is designed in such a way that oil separation is also possible in addition to the requirements of stationary operation. The valve is in this case actuated at regular time intervals for a few seconds so as to cause the oil which has accumulated in the compression space to flow out. So that a lifting off of the roller from

the eccentric of the positive drive is avoided, the rotational speed of the textile machine has to be reduced during this action (what is known as the care cycle). At creep speed, said valve is likewise opened, so that the pressure in the cylinder chamber does not rise appreciably above the feed pressure. The required power of the motor is thereby reduced, which is necessary so that the main motor can rotate at low rotational speeds and therefore manual rotation on the hand wheel is possible without excessive effort.

The disadvantage of the above solution is the high outlay for the electrical/pneumatic activation of the valve. The entire control of the pneumatic drive of the thread control device therefore has a large number of components, such as non-return valves, excess pressure valves, pressure reducing valves, and also electronic control units which make the system more susceptible to faults. Moreover, the efficiency of the textile machine is reduced as a result of the repetitive lowering of the motor rotational speed in order to discharge the lubricating oil, this lowering taking place every 15 minutes. Furthermore, this lowering of the motor rotational speed may have an adverse influence on weaving quality, for example may lead to a slight change in the width of the cloth web produced.

Presentation of the Invention

The object of the invention is to improve a thread control device of the type that has been mentioned initially.

The set object is achieved by means of the characterizing features of claim 1. Since the valve has a first valve seat connected to a cylinder chamber, and has a second valve seat, between which a valve member provided with at least one throttle point and

prestressed against the first valve seat by means of a spring is moveable, the throttle point being inactive and the valve member shutting off communication with the compressed gas source when the valve member is against the second valve seat, the valve can operate in various operating states without external activation. Furthermore, reliable oil separation is ensured, without additional measures, by the independently operating valve, without a lowering of the rotational speed, a reduction of the maximum compression pressure in the cylinder chamber under part load and a lowering of the compression pressure to the feed pressure at creep speed.

Advantageous refinements of the invention are described in claims 2 to 19.

In principle, the most diverse embodiments to the valve designed with two valve seats may be envisaged. A refinement as claimed in claims 2 and 3 is advantageous, according to which the housing has two parts, one part having at one end the first valve seat and the other part being designed as a closing-off part of the housing with a second valve seat and with a passage duct. The valve therefore has as simple a construction as possible, which allows cost-effective production and simple assembly of the valve.

The valve housing may, in principle, have various forms, a cylindrical design of the housing according to claim 4 being advantageous. This design allows a good guidance of the piston-like valve member in the housing. Moreover, the piston-like valve member may be provided with a sealing ring in order to seal off the cylinder chamber outwardly. In the version according to claim 4, it is advantageous to design the throttle points as throttle orifices formed on the valve member. According to claim 5, it is also conceivable to design

the valve member without a sealing ring, in which case a gap between the valve member and the housing wall may serve as a throttle point.

The valve may be arranged in a connecting line between the cylinder chamber and the feed pressure chamber. However, a direct arrangement in the cylinder of the cylinder/piston assembly according to claim 6 is advantageous. Furthermore, according to claim 7, it is advantageous to arrange the valve at a lowermost point of the cylinder. The valve can thus communicate directly with the cylinder chamber, and lubricating oil which has accumulated in the cylinder chamber can thus be led along a short path through the valve into the feed pressure chamber. Correspondingly, the closing-off part of the valve is connected directly to the feed pressure chamber according to claim 8, in order, again, to minimize the flow resistance and the flow path of the out-flowing oil.

The feed pressure chamber may, in principle, be of any desired design. A design as claimed in claims 9 to 12 is advantageous, according to which the feed pressure chamber may be designed with an oil separation outlet arranged at its bottom and according to which a connection for compressed air may be arranged, at a distance from the bottom of the feed pressure chamber, on a lateral wall. This arrangement of a compressed air connection and oil separation outlet prevents oil which has accumulated in the feed pressure chamber from blocking the compressed air connection or from flowing in in a connecting line of the compressed air connection. In principle, any return device may have a separate feed pressure chamber. It is advantageous, however, according to claim 12, to connect a plurality of return devices to one feed pressure chamber. A simple construction with only one connection for

compressed air and with only one oil separation outlet for a plurality of return devices is thereby possible.

In principle, the most diverse designs of the pneumatic return device according to the invention may be envisaged. In claims 13 to 16, a particularly simple design of the valve is described, in which, in conjunction with claims 5 and 6, the valve may be arranged at a lower point of the cylinder chamber of the cylinder/piston assembly. According to claim 13, a lower portion of the cylinder may serve as a housing for the valve. The valve space may advantageously be delimited by the cylinder inner face, by a closing-off part closing off the cylinder chamber and by a valve member and be connected directly to a compressed gas source via a connection arranged on the cylinder wall. A first valve seat for the valve member may be formed, according to claim 14, on an annular stop. According to claim 15, a second valve seat may be formed on a sleeve part of the closing-off part. When the valve member moves against the second valve seat, the communication of the cylinder chamber with the compressed gas source is shut off and the throttle points on the valve member become inactive. Moreover, it is particularly advantageous, according to claim 16, to arrange an oil separation outlet directly on the closing-off part.

The valve is activated as soon as the pressure in the feed pressure chamber overshoots the switching pressure. The latter depends both on the pressure in the feed pressure chamber and on the prestressing force of the spring. A refinement as claimed in claims 17 and 18 is advantageous, according to which the prestressing force can be set from outside, for example, via a screw.

The maximum compression pressure of the valve can be set, according to claim 19, by means of the flow cross

section of the throttle point. If a higher compression pressure is required, the flow cross section of the throttle point is reduced. Owing to the smaller throttle area, communication between the cylinder chamber and the compressed gas source is interrupted earlier, thus achieving a higher maximum compression pressure.

By means of the versions according to claims 17 to 19, the switching pressure and the maximum compression pressure in the cylinder chamber can be set in a simple way.

Brief Description of the Drawings

Exemplary embodiments of the thread control device of the invention are described in more detail below, for a needle-type ribbon weaving machine, by means of the drawings, in which:

figure 1 shows a needle-type ribbon weaving machine in a side view;

figure 2 shows a heald frame device with pneumatic return device in a view transverse to the running direction of the warp threads;

figure 3 shows the pneumatic return device illustrated in figure 2 as a detail and on a larger scale in the basic position;

figure 4 shows the pneumatic return device illustrated in figure 3 in the compression position;

figure 5 shows a further exemplary embodiment of a pneumatic return device on a larger scale;

figure 6 shows the pneumatic return device illustrated in figure 5 in the compression position;

figure 7a shows pressure and piston profiles of the pneumatic return device according to the invention at creep speed;

figure 7b shows pressure and piston profiles of the pneumatic return device under part load; and

figure 7c shows pressure and piston profiles of the pneumatic return device under full load.

Ways of Implementing the Invention

Figure 1 shows a needle-type ribbon weaving machine with a machine stand 2, in which is mounted a main drive shaft 4 which drives at least one weft needle 6, not described in any more detail, a reed 7, a cloth take-up 8 and a thread control device formed as a heald frame device 10. The needle-type ribbon weaving machine has a warp beam stand 12 carrying warp beams 14, from which warp threads 16 are supplied to the heald frame device 10 which opens the warp threads to form a shed 18. By means of a thread supply device 20, a weft thread 24 is supplied from a thread bobbin 22 to the weft needle 6 which introduces a weft thread loop into the shed 18. Successive weft thread loops can be tied off with themselves or by means of a tucking thread 26 which is supplied via a further thread supply device 28 to a knitting needle, not illustrated in any more detail here, in order to tie off and secure an inserted weft thread loop.

Figure 2 shows the heald frame device 10, in which a plurality of heald frames 30 with thread guide members 31 are connected in each case by means of a link 32, on the one hand, via a positive drive 35, to a cam drive 34 and, on the other hand, to a pneumatic return device 36. The cam drive 34 has pivoting levers 38 which cooperate at a drive point 40 with cams 42 of a camshaft 44. At the output point 46, the pivoting levers 38 are articulated on the links 32 via joints

48. The pivot axes defined by the joints 48 run at right angles with respect to the planes spanned by the heald frames 30. The distances A of the pivoting levers 38 of the drive points 40 from the respective pivot axes 50 are different between adjacent pivoting levers, the distances B of the output points 46 from the fixed pivot axes 50 also being different, such that, overall, the heald frames are displaceable over extents of different size, in order to form a shed continuously widening and narrowing again, as may be gathered from figure 1. The pneumatic return device 36 is formed by a cylinder chamber 52, in which a piston 54 is displaceable, which is connected to the link 32, in order to compress the piston positively at the working frequency of the cam drive 34. The cylinder chamber 52 is connected to a valve 56. The latter is preceded by a feed pressure chamber 58, via which a compressed gas source 60 is connected, in order to maintain the gas pressure in the cylinder chamber 52.

Figure 3 and figure 4 show the pneumatic return device on a larger scale during a compression action. In this case, figure 3 illustrates the piston 54 at a top dead center 66, and figure 4 illustrates the piston 54 at a bottom dead center 68 in a cylinder 64 after compression. The valve housing consists of two parts, a sleeve-like housing 70 with a first valve seat 72, formed at one end and connected to the cylinder chamber 52, and a closing-off part 74 which has a second valve seat 76 and a passage duct 78. The latter is connected to the feed pressure chamber 58. A valve member 82 provided with throttle points 80 is arranged moveably between the valve seats.

In the initial state shown in figure 3, the valve member 82 is prestressed against the first valve seat 72 by means of the prestressing force of the spring 84, so that the cylinder chamber 52 and the feed pressure

chamber 58 are in communicating connection with one another via the throttle points 80 in the valve member 82 and the passage duct 78 of the closing-off part 74. In the case of a high pressure in the cylinder chamber 52, the valve member 82 moves against the second valve seat 76 and interrupts communication between the cylinder chamber 52 and the feed pressure chamber 58, as illustrated in figure 4. The throttle points 80 are inactive in this position.

The compression/expansion action of the cylinder/piston assembly is described below by means of figures 3 and 4 and in conjunction with the graphs of figures 7a, 7b and 7c. In the latter, H stands for the stroke of the piston of the cylinder/piston assembly, with UT as bottom dead center and OT as top dead center, and PK stands for the pressure of the gas in the cylinder chamber. PS represents the necessary switching pressure so that the valve member switches from the first valve seat to the second or from the second valve seat to the first. The switching pressure PS can be divided into the feed pressure PD of the compressed gas source and the corresponding pressure PF of the spring force. VZ in this case illustrates the position of the shut-off valve and VO illustrates the position of the valve communicating with the cylinder chamber via the throttle points.

First, the piston 54 moves in the cylinder 64 from the top downward and at the same time, in a first phase, displaces air through the throttle points 80 formed on the piston-like valve member 82, toward the feed pressure chamber 58. As the piston speed increases, the pressure difference (PK - PD) across the valve member 82 rises, until the switching force generated by the cylinder chamber pressure PK on the valve member 82 overcomes the prestressing force of the spring 84 and the force on the valve member 82 generated by the feed

pressure PD, and presses the valve member 82 against the second valve seat 76. The throttle point 80 of the valve member 82 is then no longer active. By the piston 54 being moved further toward the valve 56, therefore, the cylinder chamber pressure PK rises sharply during the compression action in the cylinder chamber 52 and reaches its maximum at bottom dead center UT. In the expansion phase, the valve member 80 moves from the second to the first valve seat 76 as soon as the spring force overshoots the force generated on the valve member 80 as a result of the pressure difference (PK - PD). At the end of the expansion phase, corresponding to the top dead center 66 of the piston, the feed pressure PD is established in the cylinder chamber. Moreover, any oil which has accumulated in the cylinder chamber 52 can then flow out through the passage duct 78. During the next compression action, the out-flowing oil is blown out by the air displaced into the feed pressure chamber 58 and flows out in an oil separation outlet 88 formed on a bottom 86 of the feed pressure chamber. A connection 90 for compressed air is arranged on a lateral wall 92 of the feed pressure chamber and consequently prevents a further backflow of the oil.

Figure 5 and figure 6 show a further design variant of a pneumatic return device on a larger scale during a compression action. In this case, figure 5 again illustrates the piston 54 at a top dead center 66, and figure 6 illustrates the piston 54 at a bottom dead center 68 in the cylinder 64 after compression of the cylinder chamber 52. A valve 56a is again arranged directly at the lowermost point of the cylinder 64. The wall of the cylinder in this case serves as a valve housing, and a valve space 94 is delimited by the wall of the cylinder 64, a closing-off part 74a closing off the cylinder 64, and a piston-like valve member 82a. A stop 71 designed as a ring is arranged directly inside the cylinder 64 of the cylinder/piston assembly and

serves as a first valve seat 72a for the piston-like valve member 82a. The latter is again prestressed against the first valve seat 72a by means of a spring 84a. The spring 84a is in this case supported on the closing-off part 74a which closes off the cylinder and has an inner sleeve part 96 for guiding the spring 84a and the free end of which serves, moreover, as a second valve seat 76a for the valve member 82a. When the latter butts against the second valve seat 76a, throttle points 80a formed in the valve member 82a become inactive. Likewise, in this position, a connection 90a, arranged on the cylinder, for a compressed gas source 60 is shut off by means of the valve member 82a. Oil which has accumulated in the cylinder chamber 52 can flow out via an oil separation outlet 88a formed on the closing-off part 74a.

In the initial state shown in figure 5, the valve member 82a is prestressed against the first valve seat 72a by means of the prestressing force of the spring 84a, so that the cylinder chamber 52 is connected to a compressed gas source via the throttle points 80a in the valve member 82a. In the case of a high pressure in the cylinder chamber 52, the valve member 82a moves against the second valve seat 76a and interrupts communication between the cylinder chamber 52 and the compressed gas source 60 by shutting off the connection 90a arranged in the cylinder wall, as illustrated in figure 6. The throttle points 80a are inactive in this position.

At the end of an expansion phase, feed pressure is established in the cylinder chamber 52. Any oil which has accumulated in the cylinder chamber 52 can then flow out into the valve space 94 through the throttle points 80a. During the next compression action, the out-flowing oil is blown out by the air displaced into the valve space 94 and flows out in the oil separation

outlet 88a formed on a bottom 98 at the closing-off part 74a. The connection 90a for compressed air is arranged, at a distance from the bottom of the closing-off part, on a wall 100 of the cylinder and consequently prevents a further backflow of the oil.

Figures 7a, 7b and 7c illustrate the pressure and piston profiles of the return device according to the invention over two load cycles at creep speed for a speed of 800 rev/min (figure 7a), for part load at 1000 rev/min (figure 7b) and for full load at 4000 rev/min (figure 7c).

At creep speed to an operating speed of, for example, 800 rev/min (figure 7a), continuous pressure compensation takes place via the throttle points of the valve member, so that the cylinder pressure PK does not reach the switching pressure PS necessary for interrupting communication between the cylinder chamber and the compressed gas source. The pressure in the cylinder chamber PK is therefore always of the order of magnitude of the feed pressure PD. The motor load occurring due to the pneumatic drive is consequently low and allows the motor to run quietly and, particularly with the drive switched off, a movement of the thread control device by hand, for example for setting and repair purposes.

Under part load at 1000 rev/min (figure 7b), the cylinder chamber pressure PK reaches the necessary switching pressure PS during a cycle, whereupon the valve shuts off communication of the compressed gas source with the cylinder chamber and commences compression in the closed-off cylinder chamber. The compression of the cylinder chamber reaches its maximum at a bottom dead center UT. During the subsequent expansion, the cylinder chamber pressure PK falls below the switching pressure PS again. The cylinder chamber

is then connected once more to the compressed gas source, and, when a top dead center OT of the piston is reached, the feed pressure PD is established once again in the cylinder chamber. The compression pressure in the cylinder chamber prevents the roller from being lifted off from the eccentric of the positive drive at higher operating speeds.

Under full load at 4000 rev/min the necessary switching pressure PS is reached earlier (figure 7c) than at lower operating speeds. Compression therefore takes place over a larger stroke, and the maximum compression pressure consequently reaches a higher value than at lower operating speeds. During the subsequent expansion, the necessary switching pressure PS is reached again, whereupon the valve restores the communication of the cylinder chamber with the compressed gas source. The maximum compression pressure is a direct function of the speed of the machine, that is to say, at a higher speed, the maximum compression pressure also increases. This is advantageous both for an efficient operation of the machine and for a satisfactory functioning of the positive drive.

By the valve being opened once per work cycle, a continuous outflow of the lubricating oil which has accumulated in the cylinder chamber takes place. A reliable and continuous operation of the plant is consequently possible, without any maintenance cycles for removing the lubricating oil from the cylinder chamber. The tasks and requirements for the valve which are described above take place independently, that is to say without any external activation. The dimensioning of the spring force, of the throttle cross section and of the valve member outside diameter or valve seat diameters affords the independent control functions of the valve.

The return device described here for a thread control device consequently fulfils the most diverse requirements independently and at the same time has the least possible outlay in technical terms. The return device can therefore be produced particularly cost-effectively and, owing to its simple construction, is largely maintenance-free and fault-free during operation.

The thread control device according to the invention may also be used for individual thread control, for example for a Jacquard machine, furthermore, in a weft thread device for the presentation of individual weft threads.

List of Reference Symbols

2	Machine stand	56a	Valve
4	Main drive shaft	58	Feed pressure chamber
6	Weft needle	60	Compressed gas source
7	Reed	64	Cylinder
8	Cloth take-up	66	Top dead center
10	Heald frame device	68	Bottom dead center
12	Warp beam stand	70	Housing
14	Warp beam	71	Stop
16	Warp thread	72	First valve seat
18	Shed	72a	First valve seat
20	Thread supply device	74	Closing-off part
22	Thread bobbin	74a	Closing-off part
24	Weft thread	76	Second valve seat
26	Tucking thread	76a	Second valve seat
28	Thread supply device	78	Passage duct
30	Heald frame	80	Throttle point
31	Thread guide member	80a	Throttle point
32	Link	82	Valve member
34	Cam drive	82a	Valve member
35	Positive drive	84	Spring
36	Return device	84a	Spring
38	Pivoting lever	86	Bottom
40	Drive point	88	Outlet
42	Cam	88a	Outlet
44	Cam shaft	90	Connection
46	Output point	90a	Connection
48	Joint	92	Wall
50	Pivot axis	94	Valve space
52	Cylinder chamber	96	Sleeve part
54	Piston	98	Bottom
56	Valve	100	Wall